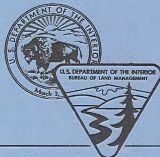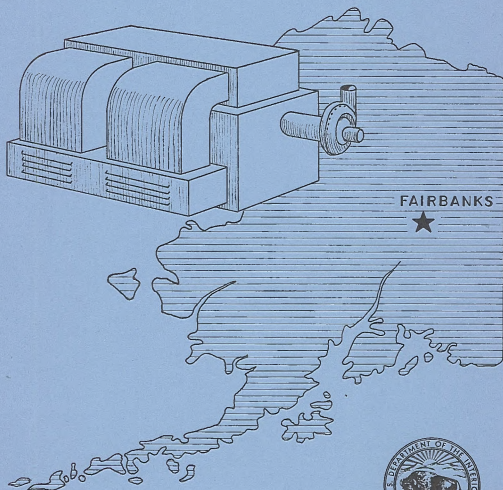# ICE FOG

ABATEMENT BY CONDENSATION

FAIRBANKS
★

U.S. DEPARTMENT OF THE INTERIOR
BUREAU OF LAND MANAGEMENT

8800435a

IAEE REPORT 7204

EVALUATION OF A FLUE-GAS TO AIR HEAT EXCHANGER
FOR PREVENTION OF ICE FOG AT FAIRBANKS, ALASKA

INTERIM REPORT

BY

A. Ronald McKay and Leory E. Leonard

April 1, 1972

Contract 52500-CT2-46
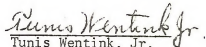Contract duration: July 1, 1971, to June 30, 1972
Contract Monitor: Mr. William C. Stringer (BLM)

Major Author:

(signature)

A. Ronald McKay,
Principal Investigator

Approved for release:

(signature)

Tunis Wentink, Jr.
Associate Director

Submitted to:

Bureau of Land Management
U.S. Department of the Interior
Denver Federal Center, Building 50
Denver, Colorado   80302

by

University of Alaska
Institute of Arctic Environmental Engineering
Fairbanks, Alaska   99701

ACKNOWLEDGEMENTS

TABLE OF CONTENTS

ABSTRACT

During each winter, cities in the interior of Alaska experience a
form of air pollution commonly referred to as ice fog. The ice fog is
frozen particles of water suspended in the air. With increased popula-
tion and activity, this condition as it applies to Fairbanks has become
progressively more acute in recent years. The formation of ice fog in
Fairbanks starts at approximately -20°F and becomes increasingly more
intense as the temperature drops.

Ice fog is produced by the introduction of water vapor into the cold
atmosphere. This water vapor is introduced in large quantities by es-
sentially two methods: 1) incomplete evaporation from open water surfaces
such as power plant cooling ponds, and 2) water vapor introduced into
the air as the result of the combustion of hydrocarbon fuels in power
plants, heating plants, and automobiles.

This project is concerned with the elimination of ice fog by removal
of the water vapor from the products of combustion which result from the
use of a hydrocarbon fuel in a heating plant. The method by which this
removal is accomplished is conceptually simple. The products of combus-
tion are cooled in a heat exchanger by the existing cold ambient air.
The products of combustion are cooled to a temperature slightly above
the freezing point of water, thus condensing the water vapor so that it
may be removed. Such a heat exchanger was installed and operated at a
Department of the Interior's Bureau of Land Management station at Fairbanks.

The results of the tests carried out show that 94 percent of the
total water vapor formed on combustion is removed by the condenser. The
condensate rate obtained at this figure showed that the exchanger removed
100 percent of the theoretically possible water vapor when operated at an
exit dew point temperature of 35°F.

-iii-

I. INTRODUCTION

A. PURPOSE OF THE PROJECT

This report represents the work carried out under Department of the Interior's Bureau of Land Management contract number 52500-CT-2-46. A copy of this contract may be found in Appendix I.

The work involved essentially consisted of the following:

1. To assist the Bureau of Land Management (the Bureau) on request, in the design and installation of an air to flue-gas heat exchanger, whose purpose is the removal of the moisture formed during combustion from the products of combustion.

2. To obtain performance data and make recommendations to assist the Bureau in the application of such a heat exchanger for eventual use on the intended Fairbanks District Office complex heating plant.

B. GENERAL

Each year, in interior Alaska, increased population and commerce make the phenomenon known as ice fog more acute. This form of air pollution is produced by frozen particles of water suspended in the air during extremely cold weather. In Fairbanks, ice fog is particularly serious due to the population density and geographical location.

Ice fog begins to form in Fairbanks at approximately -20°F and becomes more intense as the temperature drops. It is produced in large quantities in two ways: 1) incomplete evaporation above open water such as cooling ponds for power plants, etc., and 2) water introduced into the air as a result of the combustion of hydrocarbon fuels in power and heating plants and in automobiles. This project is concerned with the elimination of ice fog produced by a medium-sized heating plant. To illustrate

the visible effects, Figures 1 and 2 show the no-ice-fog and ice-fog condition on a downtown Fairbanks street.

In the following we deal with studies and tests on a heat exchanger directed toward one of several sources of the water vapor which, under the proper meteorological conditions, contribute to the ice fog encountered at Fairbanks. While the water in the Fairbanks atmosphere comes from several sources, here we treat that resulting from the fuel-oil-fired medium-sized Bureau of Land Management fire control station heating plant.

The prime function of the heat exchanger is to remove the water vapor formed on the combustion of a hydrocarbon fuel with air by cooling the products of combustion. Some knowledge of the thermodynamic and heat transfer processes involved is necessary, and this is reviewed briefly here.

The moisture which enters the heat exchanger is derived from two basic sources. First, that moisture which is formed on the primary oxidation of the hydrogen in the fuel with the oxygen contained in the air, and second, that which is contained in the air which enters the combustion process. If the water vapor is to be removed by cooling, condensation must take place in the exchanger. The products of combustion must first be cooled to the dew point of the water vapor contained in the products and then the products and the resulting condensate must be further cooled to obtain the desired water vapor extraction. The practical limit of this process is the freezing point of the condensate.

Figure 1. Downtown Fairbanks Scene (-5°F)

Figure 2. Same Fairbanks Scene as Figure 1,
With Ice Fog (-37°F)

The dew point in the exchanger, that point at which condensation first occurs, is a function of the fuel used, the amount of air supplied for combustion and the amount of water vapor carried into the system with this air. Water vapor carried over and expelled from the heat exchanger with the remaining products of combustion is a function of the three previously mentioned variables and the outlet temperature of the products from the exchanger.

The following briefly describes the various processes involved and the equations which govern them. Application of the theoretical aspects to the practical case are described by using the data of test run "C" as an example.

C. COMBUSTION PROCESS

The combustion process is shown schematically in Figure 3. For complete combustion with the theoretical air requirement and fuel components of carbon, hydrogen and sulfur the process may be represented by the following equation:

$$C_x H_y + aS + bO_2 + cN_2$$

$$\rightarrow dCO_2 + e\, SO_2 + dH_2) + cN_2$$

$$+ \text{(heat release)} \text{------------} \tag{1}$$
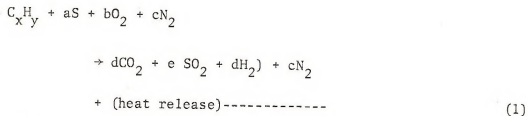
It will be noted that the oxygen requirement to balance the equation dictates the amount of air required. Thus, knowing the components making up the fuel and the air requirement, the amount of water vapor (H_2O) formed on combustion may be calculated. The quantities a, b, c, and d, etc., are the number of moles of each constituent present, and x and y are representative numbers of carbon and hydrogen molecules present in the hydrocarbon fuel.
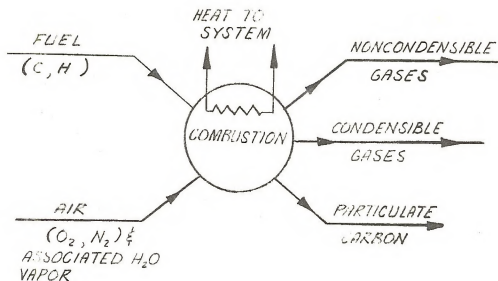
FIGURE 3

As all the fuel is not able to come into complete contact with the oxygen contained by the air required for complete combustion, an excess of air is normally supplied. In this case, equation (1) is modified as follows:

$$C_xH_y + aS + (1 + \text{% excess}) [bO_2 + cN_2]$$
$$\rightarrow dCO_2 + eSO_2 + dH_2O + (\text{% excess}) aO_2$$
$$+ (1 + \text{% excess}) cN_2 + (\text{heat release}) \tag{2}$$
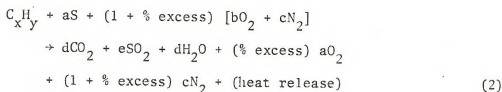
The (% excess) term is expressed as a decimal. It will be noted that the moles of carbon dioxide, sulfur dioxide, and water vapor have not changed by the inclusion of the excess air; however, the mole fraction of the various components in the products of combustion have changed.

To illustrate the above consider the data of test "C".

Fuel data: (See Appendix II, letter from Union Oil)

Element	% by Weight
Carbon	86.5%
Hydrogen	13.5 %
Sulfur	trace

-5-

On a per 100 pounds of fuel fired per hour basis and neglecting the sulfur, the representative combustion equation from (1) becomes:

$$C_{7.2}H_{13.5} + 10.575O_2 + 39.75N_2$$

$$\rightarrow 7.2\ CO_2 + 6.75\ H_2O + 39.75N_2 \qquad (3)$$
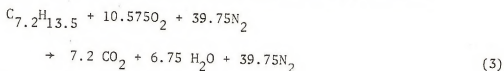
Equation (3) indicates the following on a weight basis:

Air required for 100 pounds of fuel:

O_2 =	10.575 X 32	=	338.4	pounds
N_2 =	39.75 X 28	=	1,113.0	pounds
		TOTAL =	1,451.4	pounds

products of combustion:

CO_2	7.2 X 44	316.8	pounds
H_2O	6.75 X 18	121.5	pounds
N_2	39.75 X 28	1,113.0	pounds
	TOTAL	1,551.3	pounds

These values show that for each pound of fuel consumed:

air required	14.51 pounds
CO_2 formed	3.168 pounds
N_2O formed	1.215 pounds

Consideration of equations (1) and (2) indicates that even though excess air is supplied, the water vapor content of the products does not increase. However, if one is to consider the total water vapor input to the condenser, the amount of air supplied must be accounted for because of the water vapor it carries with it. Thus, for total moisture input, the excess air must be known.

During most of the trial runs, the CO_2 content of the combustion

products at the boiler averaged 10.3 percent. Application of equation (2) yields Figure 4 in terms of CO_2 content on a percent by volume basis as a function of % excess air. From this figure, or equation (2), the 10.3 percent CO_2 content gives a % excess air of 30 percent. With this excess, the weight of air per pound of fuel consumed is now:

weight of air supplied: 18.86 pounds/pound fuel

fuel consumed per unit time   45.8 pounds/hour

total air supplied   863.79 pounds/hour

    During test run "C" the outside ambient was -16°F with incipient ice fog conditions. Ambient air may then be assumed as saturated. Though the combustion air enters the warmer heating plant and is thus warmed, its ambient water vapor content is not diminished but, in all probability, increased due to the presence of an open feed water makeup tank at the higher building temperatures. Thus, assuming the combustion air as having the ambient air water-vapor content is conservative.

    At saturation with an ambient of -16°F, each pound of air has associated with it 0.0003301 pounds of water. Then the water or water vapor associated with the air is:

Air entrained water vapor:

863.79 X 0.0003301

= 0.285 pounds/hour

    The total amount of water vapor which then enters the exchanger is:

from combustion   1.215 X 45.8 = 55.647   pounds

by combustion air   0.285   pounds

                        TOTAL   55.932   pounds/hour
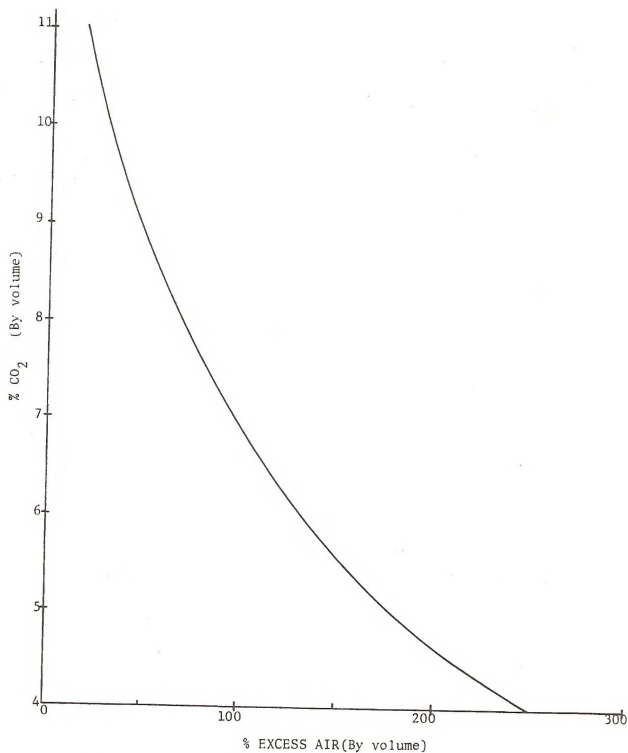
FIGURE 4. Percent CO_2 in Flue Gas versus Percent Excess Air
(For a 38.8° API Fuel Oil)

It should be noted here that a difference existed in the CO_2 content after combustion and before introduction of the products to the heat exchanger. This difference--a decrease--was due to the damper system in the normal smoke stack of the heating plant. The damper allowed additional air to be entrained. The additional air did not seriously affect the moisture addition but did affect the moisture carryover from the heat exchanger.

II. HEAT EXCHANGER SYSTEM

A. HEAT EXCHANGER REQUIREMENTS

The heat exchanger must accomplish two functions in the removal of the water vapor by condensation. It first must cool the products of combustion, including the water vapor, to the dew point of the water vapor. Diagramatically, the exchanger's function is shown in Figure 5. In this function, it is acting as a de-superheater as far as the water vapor is concerned and, at the same time, a gas cooler for the other products of combustion. Since water vapor under these conditions acts as a gas, we have essentially a gas cooler. Once the dew point has been reached for the water vapor, we have a condenser with a continuously changing dew point for the remaining water vapor as water vapor is condensed out. At the same time, we have a gas cooler for the remaining non-condensible products of combustion. The problem is somewhat further complicated if any of the non-condensibles such as SO_2, CO_2, and O_2 are dissolvable in the water vapor condensate.

What is implied here is that after the initial condensation, the dew point continuously changes as water vapor is condensed and dissolvable gases with an affinity for water are dissolved into the condensate. Considering test run "C" and the implications of equation (2), equation (3) may be reduced to give the temperature at which condensation first occurs.

The partial pressure of each component in the products is in direct relationship to the total pressure of the products and is equal to the product of its mole fraction and the total pressure.

In the condenser the pressure is essentially ambient, which for Fairbanks is 14.48 psia. Thus, considering equation (3) and with excess air,
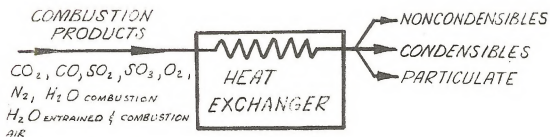
COMBUSTION PRODUCTS

CO_2, CO, SO_2, SO_3, O_2, N_2, H_2O COMBUSTION
H_2O ENTRAINED & COMBUSTION AIR

HEAT EXCHANGER

NONCONDENSIBLES
CONDENSIBLES
PARTICULATE

FIGURE 5

the partial pressure of H_2O is:

$$PH_2O = 14.48 \times \left[\frac{6.75}{7.2 + 6.75 + 3.17 + 51.68} \right]$$

$$= 1.42 \text{ psia}$$

With this vapor pressure initial condensation will occur at a temperature of approximately 113.75°F to 114°F.

Should the mole fraction of the H_2O change, by increased excess air, a different composition fuel, or increased air introduced water vapor, the dew point of initial condensation will shift accordingly.

High sulphur content fuels may raise this dew point to as high as 250°F due to the SO_2 or SO_3 dissolving in the condensed water vapor. (See Industrial Engineering Chemistry, 28, pp. 999, 1936.)

Hence, as the heat exchanger cools, as well as de-superheats and condenses, its design is unique and applicable to one fuel under a specific set of operating conditions.

If the exit conditions of the heat exchanger are fixed or known, the water vapor which is not condensed and carried over may, by extension of equation (2), be shown as a function of exit temperature and excess air. With the fuel used in test run "C", these results are presented in Figure 6. To be noted here is that very small changes in exit temperature or variations in excess air produce significant changes in the water vapor carryover. Furthermore, as the outlet temperature is limited by the freezing point of the condensate, total extraction is not possible, and the amount which can be condensed out is affected by the above variables.

Figure 7 represents the amount of water vapor present in the products of combustion after condensation has begun, with the fuel used during test run "C" and 30 percent excess air.
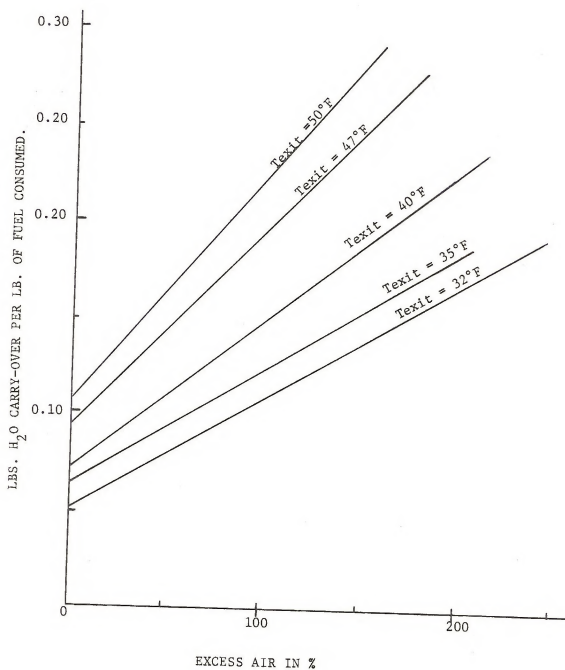
FIGURE 6. Weight of H_2O carry over versus Percent Excess
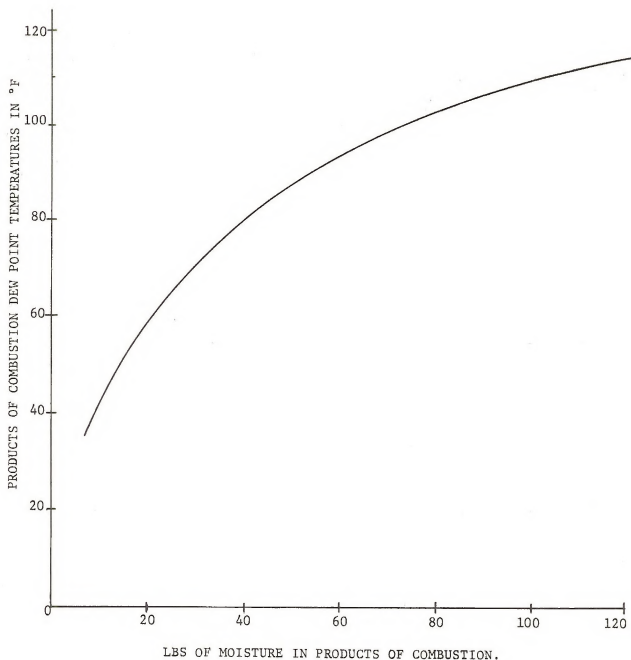Combustion Air for Various Gas Exit Temperatures
(For a 38.8° API Fuel Oil)

-13-

FIGURE 7.   Dew Point versus Weight of H_2O in Products of Combustion
            (For 100 pounds of 38.8° API Fuel Oil Burned with 30%
            Excess Air)

-14-

If the axial temperature distribution in the condenser tube is known, the amount of water vapor condensed to the point at which that temperature exists may be found. At the same time, with the total water vapor formed on combustion known, the difference between this value and that shown in Figure 7 represents the amount being carried as a vapor by the products of combustion at the temperature in question.

1. Heat Transfer Considerations

It has been previously stated the heat exchanger performs two basic functions. It first cools the products of combustion and the associated water vapor to the temperature at which the water vapor first condenses. It then acts as a condenser for the water vapor and, at the same time, a cooler for the non-condensible gases.

The heat lost in the first section may be represented as:

$$Q_{(1)} = \overset{\circ}{M} \, C_{p\,mix} (T_{in} - T_{dp})$$

where: $\overset{\circ}{M}$ is the mass flow of the products in pounds/hour, including the water vapor.

$C_{p\,mix}$ is the weight-averaged, mean specific heat of the components making up the products in BTU/ hr.lb. °F

T_{in} is the inlet temperature to the heat exchanger in °F.

T_{dp} is the temperature at which condensation of the water vapor first occurs.

The heat lost in the second section may be represented as:

$$Q_{(2)} = \overset{\circ}{M}_{nc} \, C_{pnc} (T_{dp} - Tex) + \overset{\circ}{M}_{wvc} \left(\frac{h_{fg@T_{dp}} - h_{fg@T_{ex}}}{2} \right.$$

$$+ \overset{\circ}{M}_{wv} \, C_{pw} \, \frac{(T_{dp} - T_{ex})}{2}$$

where: $\overset{\circ}{M}_{nc}$ is the mass flow of the non-condensible products.

C_{pnc} is the weight averaged mean specific heat of the non-condensibles

T_{ex} is the exit temperature of the heat exchanger.

$\overset{\circ}{M}_{wvc}$ is the amount of water vapor condensed.

C_{pw} is the specific heat of the condensate.

$h_{fg@T_{dp}}$
and
$h_{fg@T_{ex}}$ are the latent heat of vaporization at the first condensing temperature and the exit temperature, respectively.

It will be noted that the average values of the latent heat of vaporization and the average value of the temperature difference through which the condensate is cooled are used. This approximation is necessary as the latent heat is a function of the existing condensation temperature along the tube and all of the condensate is not formed at temperature T_{dp}. An exact calculation would consider the integrated effect from the dew point of first condensation to the exit temperature. This would require the known axial temperature distribution in the condenser tube.

The total heat lost by the exchanger, considering both sections, is then:

$$Q_{hx} = Q_{(1)} + Q_{(2)}$$

As an example of the above, consider test run "C":

fuel consumed	45.8 pounds/hour
air supplied	863.79 pounds/hour
air-entrained moisture	0.285 pounds/hour
products of combustion for the first section of the heat exchanger	910.2 pounds/hour

mean specific heat for products	$0.2513 \frac{BTU}{Lb.°F}$

heat exchanger inlet temperature T_{in}	225°F

dew point temperature T_{dp}	114°F

Then the heat lost in the first section of the heat exchanger:

$$Q_{(1)} = 910.2 \times 0.2518 \times (225-114)$$

$$= 25,439.9 \text{ BTU/hour}$$

For the second section of the heat exchanger:

non-condensible products of combustion	854.3 pounds/hour
mean specific heat for non-condensibles	$0.2366 \frac{BTU}{Lb.°F}$
total water vapor	55.932 pounds/hour
water vapor carried over	3.555 pounds/hour
water vapor condensed	52.377 pounds/hour
average latent heat of evaporation	$1,051.09 \frac{BTU}{Lb.}$
exit temperature T_{ex}	35°F

The heat lost by the products in the second section of the heat exchanger is then:

$$Q_{(2)} = 854.3 \times 0.2366 \times (114-35)$$

$$+ 52.377 \times 1,051.09$$

$$+ 52.377 \times 1.0 \times 39.5$$

$$= 73,089.2$$

The total heat lost is then:

$$Q_{hx} = Q_{(1)} + Q_{(2)}$$

$$= 24,439.9 + 73,089.2$$

$$= 73,089.2 \ \frac{BTU}{Hr.} \quad 97,529.1$$

B. HEAT EXCHANGER PERFORMANCE

Operating as a device which removes water vapor from the products of combustion, the performance of the heat exchanger may be expressed as follows:

$$performance = \frac{wt. \ of \ water \ actually \ condensed}{wt. \ of \ water \ which \ can \ be \ condensed \ theoretically}$$

The weight of water vapor which could theoretically be condensed is equal to that which could theoretically be formed on combustion plus that which is entrained in the combustion air, less that which is carried over with the cooled products of combustion at the exit of the heat exchanger.

For test run "C":

water vapor formed on combustion	55.647 pounds/hour
water vapor in combustion air	0.285 pounds/hour
water vapor carried over by products at exit temperature 35°F	3.555 pounds/hour
condensate as measured	54.2 pounds/hour

then:

$$performance \ \% = \frac{54.2}{55.647 + 0.285 - 3.555}$$

$$= 100 \ \%$$

It should be noted that the water vapor condensed becomes a function of the exit temperature. Complete removal of the moisture which enters the exchanger is not possible if the condensate is to remain in the liquid state.

The moisture removed as a percentage of the total introduced may be expressed as a percentage as follows:

$$\% \text{ removed} = \frac{\text{wt. condensed}}{\text{total water vapor entering condenser}}$$

$$= \frac{52.4}{55.9}$$

$$= 93.74\%$$

This value of approximately 94 percent is considered to be a maximum if no freezing is to take place in the tube.

C. ANALYSIS OF EXISTING HEAT EXCHANGER DESIGN

The design specifications of the existing design are presented in the appendix. Initially, it was expected that, though the exchanger had adequate surface area for its designed load, difficulty would be experienced in two prime areas during its operation. These areas were in the freezing of the condensate within the tubes and in the pressure drop which would result across the exchanger due to the flow of the combustion products through the tubes.

1. <u>Tube Freezing</u>

As the exchanger was to be installed outside the existing heating plant, the exterior surfaces of the tubes in the exchanger would be subjected to the lowest ambient temperature which might exist. For the Fairbanks installation, this could be lower than -60°F. If the tubes are not to freeze then the temperature on the inner surface of the tube must never be less than 32°F. The limitation imposed by the freezing temperature and that of the lowest ambient combine to place severe design restrictions on the exchanger design. These restrictions are best discribed with the aid of Figure 8 , which represents the temperature distribution across the tube wall and inner and outer gas films.
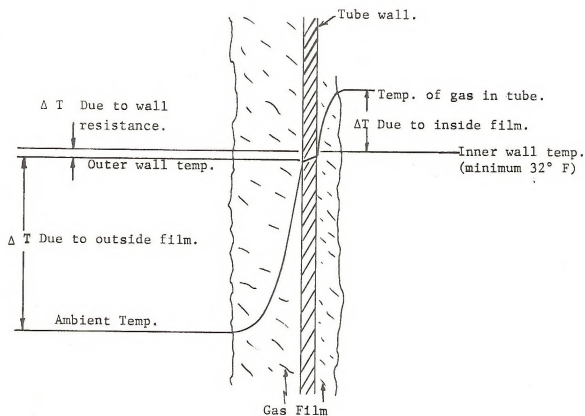
FIGURE 8. Temperature Distribution Across Heat Exchanger
Tube Wall.

Though the products of combustion within the tube may be well above the limiting freezing temperature, the water vapor is free to diffuse through the film and may come in contact with the wall. The inner wall temperature is thus limited to a minimum value of 32°F. For a given heat flow per unit area through the tube wall, the respective temperature-drops across the films and the tube wall depend upon the conductances involved. The conductances of the films which will control depend upon the gas velocities and the characteristic tube dimension involved. For flow of ambient air over the outside of the tubes, the characteristic dimension is the outer diameter, while for flow inside the tubes it is represented by the inner diameter. What is implied is that with a fixed tube and fixed internal and external velocities, there is a minimum ambient temperature below which freezing definitely takes place on the inside of the tube. This minimum ambient temperature becomes successively higher with distance down the tube as the products of combustion are cooled.

In its original design configuration, the exchanger was to operate in a start-stop mode following the cycling on and off of the boilers. During the off-mode with the inlet and outlet shutters closed, there was sufficient heat loss from the exchanger enclosure to bring the tube material to below the freezing temperature. On successive start-up, the tubes froze completely closed before they had sufficient time to warm up to the freezing point at which the condensate would be able to drain continuously.

It should be noted that, though it is common practice to allow a safety factor in heat exchanger design by increasing the required area, this is to be avoided in cases where freezing of a condensible may take place.

-21-

2. Pressure Drop

The design calculations of the H-X manufacturer for the anticipated pressure drop are included in the appendix. Probable pressure drop is given as two to three inches of H_2O at the maximum gas flow rate through the exchanger. It was also considered that natural draft utilizing a stack of sufficient height would be available to overcome the exchanger pressure drop. This latter aspect is clearly impossible, as shown in Figure 9. That figure gives the stack height necessary for a gas exit temperature of 35°F as a function of various ambient temperatures.

D. MODIFICATIONS TO EXISTING DESIGN

Certain modifications were necessary to the existing design in order that it would function in a continuous manner. The modifications were required to prevent freezing in the tubes and to overcome the pressure drop across the exchanger.

To allow sufficient steady state operating time for test purposes, one boiler was cut back to the low fire position so that it would operate with a maximum on-the-line-duty cycle. Prevention of freezing in the tubes was accomplished by insulating the inlet and outlet headers and by supplying additional heat to the airbox, creating an artificial ambient temperature. The heat was supplied by an auxiliary heater and the airbox temperature maintained constant by bleeding cold ambient air into the exchanger enclosure. Only one of the two fans was operated, so the air circulated in a loop fashion within the airbox.

The pressure drop across the tube side of the heat exchanger was overcome by using an exhaust fan at the outlet of the exchanger. An air bleed at the inlet to the exhaust fan allowed the pressure drop across
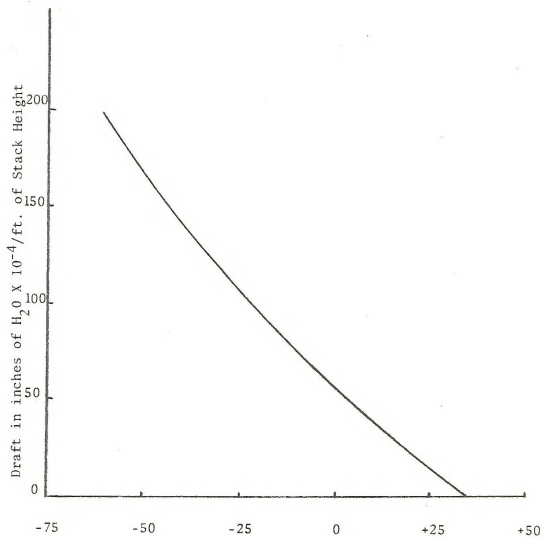
FIGURE 9. Theoretical Draft for an Exit Gas Temperature of +35°F versus Ambient Temperature.

-23-

the heat exchanger to be adjusted so that the smoke pipe at the boiler operated at between 0.05 and 0.10 inches H_2O negative pressure.

It was also necessary to construct a roof above the heat exchanger unit to keep snow from building up on the upper air louvers and freezing them shut. The modifications are shown on Figure 10, which also shows the heat exchanger in relation to the existing heating plant. Figures 11 and 12, presented later, also show the actual installation.
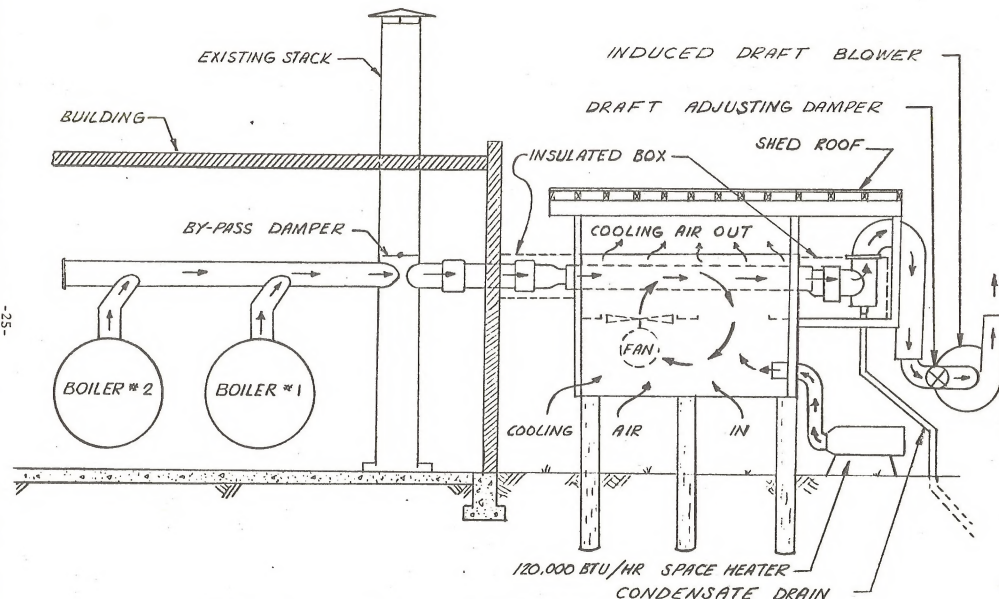
FINAL TEST CONFIGURATION

FIGURE 10

III. TEST DATA

Though the principle is simple, the operation of a heat exchanger at temperatures close to the freezing temperatures of water, necessary to condense out the major portion of water vapor in the exhaust gas, presents many engineering problems. There are two aspects to this project: 1) reduction of visible ice fog produced by oil-fired heating plant at the Bureau of Land Management fire control station at Fairbanks, and 2) develop the engineering design criteria necessary to utilize this ice fog prevention system in a practical and permanent installation.

A. VISUAL EFFECTS

Results of the visual study are best illustrated by Figures 11 and 12. Figure 11 shows water vapor produced at the Bureau of Land Management's heating plant without the condensing heat exchanger in operation. The ambient temperature during this test was -30°F. Figure 12, under the same ambient conditions, was taken some twenty minutes after Figure 11. In this case, with the exchanger operating, the extent of reduction of the visible water vapor cloud is evident. This reproducible effect was noted in many tests made over the ambient temperature range of +20°F to -49°F.
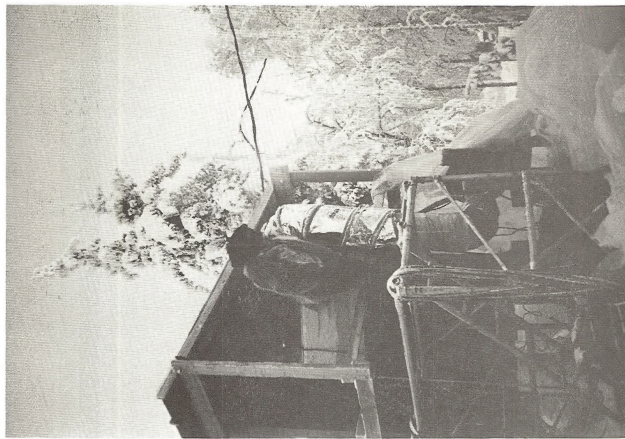
Figure 12. Heat Exchanger In Operation

Figure 11. Heat Exchanger Not In Operation

B. TEST DATA

The data obtained from the series of tests are presented in tables A through C. Included in these tables are the analyses of the condensate obtained for certain test runs. Though the depth of this analytical approach was not part of the original program, it was considered necessary to assess the pollution potential of the resulting condensed water vapor. No comment on this analysis is made other than the fact that the condensed water vapor acts as a scrubbing agent and dissolves into solution a portion of the soluble products of combustion.

Complete calculations have been presented in the text for the case of test run "C". This run was considered to be the most representative of the runs conducted at the design exit temperature of the heat exchanger. These calculations show that there is very close agreement between the performance obtained and that predicted by the theoretical aspects.

The data on runs other than test run "C" are presented as a matter of interest only, as complete sets of data were not available on many of these runs.

TABLE A

GENERAL DATA SUMMARY

Run	Ambient Air Temp. Degrees F	Cooling Mode	Inlet Gas Temp °F	Exit Gas Temp °F	Air Box Temp °F	Press of Stack ins. H_2O	Δ-Press H-X ins. H_2O	Fuel Rate Lbs./Hr.	Condensate Rate Lbs./Hr.	Remarks
1/13/72	-20	Forced Convection	218	35	17.0	--	1.5	--	--	
1/14/72	-36	"	198	35	20.0	-0.25	1.0	--	--	
B	-19	Free Convection	220	40	-14.0	-0.20	0.6	46.0	40.8	
C	-16	Forced Convection	250	35	25.0	-0.05	0.4	45.8	52.4	
D	+13	"	210	47	38.0	-0.10	0.4	46.5	45.2	
E	+14	"	220	30	14.0	-0.10	0.4	47.0	40.8	
F	+29	"	215	40	29.0	-0.05	0.4	49.6	50.0	
G	-24	"	275	35	23.5	-0.05	0.35	--	--	
H	-20	"	250	35	28.0	-0.05	0.35	--	--	
J	-20	"	260	37	26.0	-0.05	0.35	--	--	
K	+ 4	"	280	52	44.0	-0.10	0.7	--	--	
L	+ 4	"	280	52	44.0	-0.10	0.7	--	--	
M	+ 4	"	280	52	44.0	-0.10	0.7	--	--	
N	- 4	"	225	35	22.0	-0.10	1.1	47.2	58.8	

TABLE B

GAS DATA SUMMARY

Run	CO_2 @ Boiler in % by Vol.	CO_2 @ H-X Inlet % by Vol.	CO_2 @ H-X Exit % by Vol.	SO_2 @ H-X Exit ℓ/ℓ of air	CO @ H-X Inlet in P.P.M.	CO @ H-X Exit in P.P.M.	Remarks
J	10.3	9.6	9.6	--	--	--	
K	--	--	--	167.0×10^{-9}	13.0	--	
L	--	--	--	124.0×10^{-9}	--	--	
M	7.7	6.0	5.7	--	18.5	18.0	
N	6.0	5.0	4.2	--	--	--	

TABLE C

CONDENSATE DATA SUMMARY

Run	pH	NO_2-N (mg/ℓ)	NO_3-NO_2 (mg/ℓ)	$SO_4^=$ mg/ℓ	NH_3 (mg/ℓ)	Remarks
B	3.7	0.08	0.05	640	--	
C	5.7	1.7	1.10	690	--	
D	5.8	0.04	0.09	700	--	
E	5.8	0.05	0.12	620	--	
F	3.2	--	--	--	--	
J	5.6	--	--	--	--	
K#1	2.9	--	0.19	360	5	
K#2	3.1	--	0.20	440	7	
N	3.3	--	1.61	730	21	
1/16/72	4.8	0.016	.31	345	2.6	

IV. RECOMMENDATIONS

The project was successful in accomplishing its objectives of removal of up to 94 percent of the water vapor formed on combustion from the products. However, it is recommended that additional information be obtained from the existing installation. This additional information is considered necessary before the design of a unit for the intended new facility is attempted.

The present heat exchanger unit was never operated continuously or unattended, as would be necessary in a permanent installation. This was primarily due to the cycling of the boilers, on and off, and the effect of the off period on the freezing of the tubes. Furthermore, it was necessary to supply additional heat to the exchanger enclosure so that the tubes would not freeze during normal operation. It was also necessary to supply a negative pressure across the exchanger by use of an exhaust blower to overcome the excessive pressure drop.

It is suggested that a series of tests be conducted to evaluate the possibility of continuous operation of the heat exchanger unit. In this respect the following modifications to the existing unit and possible tests are recommended.

1. Adapt the existing boiler units to a modulated fuel rate with load demand rather than an on-off operating cycle.

2. Install air boxes or plenum chambers over both the existing inlet and outlet air louvers. The plenum chambers would have variable orifices to vary the amount of cold ambient air entering the exchanger enclosure, and would be controlled by the exit temperature of the heat exchanger.

3. The entire heat exchanger enclosure and plenum chamber be insulated with sufficient insulation material in such a way that the external heat loss from the assembly would not be greater than the amount of heat which the products of combustion are able to supply when the boiler is on its lowest firing rate.

4. Modify the inlet and outlet headers to minimize the pressure drop across the exchanger as well as afford a better distribution of gas flow through the tubes in the tube bundle.

5. Install an exhaust fan unit at the exit of the exchanger whose flow rate and suction pressure would match and be controlled by the requirements imposed by the variable fuel firing rate.

6. Instrumentation of several tubes with temperature sensors such that the axial temperature distribution within the tubes may be found.

7. Conduct a feasibility study on the possible use of a heat exchanger system which does not require a large power requirement to overcome pressure drop or circulate cooling air.

8. Conduct a feasibility study on the possible recovery of the heat from the products of combustion and the possible economic trade-offs.

9. Conduct a study of the recycling of the condensate effluent as boiler makeup or service water.

10. Conduct tests on the modified unit to obtain a complete heat balance and weight balance of all the elements which enter the combustion process and leave either as exit gases or condensate effluent.

11. After several hundred hours of operation, remove and section tube samples to assess corrosive aspects.

12. Introduce steam or water into the flue gas to deter-
    mine if additional pollutants can be removed by the
    scrubbing action.

V. CONCLUSIONS

The results of this study have shown that:

1. Ice fog production by hydrocarbon combustion emissions
   can be significantly reduced by utilizing a vapor condens-
   ing heat exchanger in the exhaust system to remove the
   water formed on combustion before the gases are emitted
   to the atmosphere.

2. The engineering design of heat exchangers for this pur-
   pose and the related control equipment are quite
   complex. Although a great deal of information necessary
   for the design of a continuously operational system is
   now known, more is required and continued study is con-
   sidered necessary.

3. Further research is necessary if a serious effort is
   going to be made to reduce the form of air pollution
   known as ice fog in Arctic and Subarctic regions.

APPENDIX I

CONTRACT FOR RESEARCH BETWEEN

UNIVERSITY OF ALASKA AND BUREAU OF LAND MANAGEMENT

Contract 52500-CT2-46

CONTRACT NO. 52500 - CT2 - 46

BETWEEN THE

BUREAU OF LAND MANAGEMENT
UNITED STATES DEPARTMENT OF THE INTERIOR

AND

UNIVERSITY OF ALASKA
THE INSTITUTE OF ARCTIC ENVIRONMENTAL ENGINEERING

THIS CONTRACT is entered into and effective this first day of July, 1971, by and between the Bureau of Land Management, U. S. Department of the Interior (hereinafter also referred to as "The Bureau" or "The Government") and the University of Alaska, Institute of Arctic Environmental Engineering (hereinafter also referred to as "U of A" or "The Contractor").

WHEREAS the above parties have previously entered into a Basic Agreement dated July 1, 1971, and

WHEREAS the parties desire to enter into a contract for performance by the Contractor of the below described study during the period July 1, 1971, - June 30, 1972,

NOW, THEREFORE, IT IS AGREED THAT:

1. INCORPORATION OF BASIC AGREEMENT. The Basic Agreement previously executed by the parties on July 1, 1971, is hereby incorporated by reference into this contract.

2. TITLE OF STUDY. The title of the required study is "Monitoring of Design and Performance of Air-to-Flue Gas Heat Exchanger for the Prevention of Ice Fog at Fairbanks, Alaska."

3. CONTRACT PERIOD. This study shall commence on July 1, 1971, and end on June 30, 1972, unless it is completed earlier, or renewed, or terminated as set forth below. Annual progress reports will be submitted by March 1 in lieu of semi-annual progress reports referred to in the Basic Agreement.

4. RENEWAL. It may be desirable to continue the study beyond the period set forth above. Accordingly, this contract may be renewed for additional one-year periods, each forthcoming July 1, if agreeable to both parties. Such renewals shall be in the form of written bilateral instruments. The study is initially planned for 2 years' duration.

5. TERMINATION OF CONTRACT. The Contracting Officer, by 30 days written notice, may terminate this contract, in whole or in part, when

-36-

it is in the best interest of the Government to do so. To the extent
that the contract is for services and is so terminated, the Government
shall be liable only for payment, in accordance with the payment pro-
visions of the contract, for services rendered prior to the effective
date of termination. The Contractor may also terminate the contract by
giving 30 days written notice and the Government shall only be liable
for payment as set forth above.

6. LIABILITY FOR PAYMENTS AND CONTINGENCY. The liability of the
Government for payments based on obligations incurring on or after
July 1, 1971, is made contingent upon the Congress of the United States
appropriating the necessary funds to cover such obligations. The
Government will not be liable for any payments hereunder, arising from
obligations incurring on or after July 1, 1972, unless the contract is
renewed, as set forth in Clause 4 above.

7. LOCATION OF STUDY. Fairbanks and College, Alaska.

8. JUSTIFICATION. The Bureau of Land Management presently has an
office complex at 3-1/2 Mile Airport Way, Fairbanks, Alaska. The com-
plex is equipped with a heating plant. Furthermore, the Bureau is
preparing to start the design phase for a new proposed complex to be
designated the Fairbanks District Office Complex. Before the design
phase is completed, the Bureau must obtain positive design information
on the handling of flue gases without causing ice fog. Accordingly, the
present heating plant will be used as a research vehicle for the air-to-
air heat exchanger developed as a result of this and other contracts.

    The Fairbanks-Ft. Wainwright area of interior Alaska has a form of
air pollution known as ice fog. Ice fog is produced when stable pressure
systems produce prolonged, extreme (below -35°C) cold spells at the sur-
face. The meteorological conditions responsible for the cold spells
show that the rate of radiative cooling of the air is enhanced by sus-
pended ice crystals which are themselves a result of the initial cooling.
Radiation fogs formed during the onset of cold spells are generally of
short duration because the air soon becomes desiccated. These fogs con-
sist of supercooled water droplets until the air temperature goes below
the "spontaneous freezing point" for water droplets (about -35°C); the
fog then becomes an ice crystal fog, or simply "ice fog." During the
cooling cycle,water is gradually condensed out of the air until the drop-
lets freeze. At this point there is a sharp, discontinuous decrease in
the saturation vapor pressure of the air because it must be reckoned
over ice rather than over water. The polluted air over Fairbanks allows
droplets to begin freezing at the relatively high temperature of -35°C.
This results in rapid and widespread formation of ice fog which persists
in the Fairbanks area as long as the cold spell lasts. The persistence
of Fairbanks ice fog depends on a continual source of moisture (4.1×10^6
kg H_2O per day) from urban environmental emissions within the fog. Ice

fog crystals are an order of magnitude (5 - 10 microns) smaller than
diamond dust, or cirrus cloud crystals, which in turn are an order of
magnitude smaller than common snow crystals (0.01, 0.1 and 1 to 5 mm
respectively). The differences in size are shown to result from
differences in cooling rates over five orders of magnitude. Most of the
ice fog crystals have settling rates which are slower than the upward
velocity of air over the city center. The upward air movement is caused
by convection cells driven by the 6°C "heat island" over Fairbanks. This
causes a reduced precipitation rate which permits the density of ice fog
in the city center to be three times greater than that in the outlying
areas. The inversions which occur during cold spells over Fairbanks
begin at ground level and are among the strongest and most persistent
in the world. They are three times stronger than those in the inversion
layer over Los Angeles. Thus, the low-lying air over Fairbanks stagnates
and becomes effectively decoupled from the atmosphere above, permitting
high concentrations of all pollutants.

9.   <u>OBJECTIVES</u>. The objectives of this contract are:

A. To assist the Bureau in the design, installation, and to
    monitor the operation of one (1) air-to-air heat exchanger to
    be used in a pilot type application at Fairbanks, Alaska, for
    reasons stated previously.

    The intent is to cool the flue gases with outside air to a
    point where condensation is eliminated or reduced to an
    acceptable level. Levels of acceptance have not yet been
    established.

    The unit shall be designed with a tee and damper section in
    the existing stack to allow bypassing the exchanger. It is
    envisioned that the unit be mounted on structural steel mem-
    bers and installed on a suitable support system adjacent to
    the power plant. The support system will be Government
    designed and installed with any recommendations the Contractor
    makes incorporated. Due to the unavailability of many materials
    in the Fairbanks area, the unit shall be fabricated as a
    package unit requiring only piping, power, welding, and so
    forth, required on installation. All appurtenances not attached
    to the unit shall be furnished and shipped with the unit. Air
    for pneumatic type controls is not available in the power plant.
    The boilers are automatic fire and do not operate continuously.
    The firing control is a Fireye and the boiler starts on low
    fire and advances to high fire. A time delay shall be included
    in the exchanger control system to allow the boiler fire to
    stabilize prior to starting the heat exchanger unit.

Provisions shall be made in the inlet breeching to the exchanger for introducing outside air should it be required for tempering prior to entering the exchanger.

B. To obtain performance data and make recommendations to further assist the Bureau in the eventual construction of the heating plant of the intended Fairbanks District Office Complex.

10. <u>WORK TO BE PERFORMED UNDER THIS CONTRACT</u>.

A. U of A will assist upon request of the Bureau in proposal evaluation related to design, fabrication and install- ation of the air-to-flue gas heat exchanger.

U of A will at the time of acceptance of design and prior to fabrication inform the selected contractor where any special connections, fittings, and so forth, will be required to accommodate their (IAEE) test equipment.

B. U of A (IAEE) will provide instrumentation for collecting data, compile results and make recommendations as to feasibility and possible future modifications.

Emphasis will be on H_2O emitted from the stack, but SO_2, CO_2, CO, O_2, and N_2 will also be studied. Every effort will be made to obtain baseline data before the seasonal start of ice fog formation. The ice fog behavior as a function of the heat exchanger operation will be recorded when possible photographically. This is in addition to quantitative analysis of exit gases by various instrumental means.

11. <u>PROGRAM ORGANIZATION AND MISCELLANEOUS</u>. The Principal Investigator and Program Manager will be Professor A. R. McKay, of the Department of Mechanical Engineering and the Institute of Arctic Environmental Engin- eering. He will be responsible for program planning and direction. Appropriate and qualified graduate assistants will also be employed.

A. <u>Travel</u>. Some local travel funds are budgeted in view of anticipated repetitive travel between the Bureau site and the University campus. Two round trips, one each to Denver, Colorado, and Houston, Texas, are also planned, based on conversations with Bureau personnel.

B. <u>Capital Equipment</u>. Some needed capital equipment not allowed by the Bureau regulation will be supplied at no charge by the University of Alaska; for example,

thermocouple potentiometer, gas analysis equipment.
Associated labor charges are noted under analytical
services in the cost estimate.

12. COST ESTIMATE. The cost estimate is given below. A cost reimburse-
ment type contract with predetermined fixed rate for indirect costs is
intended.

APPENDIX II

LETTER FROM UNION OIL COMPANY

UNI⑦n

H. C. Huffman
Associate Director in charge of
Refining and Products Research Division

Cloyd P. Reng
Manager Refining Research

R. J. Askevold
Manager Products Research

March 14, 1972   PROD-72-331

Mr. A. Ron McKay
Institute of Arctic Environmental
    Engineering
University of Alaska
College, Alaska 99701

Dear Mr. McKay:

HYDROGEN CONTENT OF UNION
ALASKA STOVE OIL

In line with your telephone conversation with Mr. E. L. Wiseman
we have determined experimentally the hydrogen content of a
sample of Union Alaska Stove Oil from the holding tank at the
Bureau of Land Management Fire Control Station heating plant at
Fairbanks. The sample had an API gravity of 38.8° at 60°F (or
specific gravity at 60°/60°F of 0.8309). The experimentally
determined hydrogen content was 13.51% (±0.02%) by weight.
Assuming all hydrogen present in the fuel is oxidized to water
we would expect the formation of about 1.21 pounds of water vapor
for every pound of fuel burned. In terms of volume this would
be approximately 8.35 lb of H_2O/gal of fuel at 60°F or 8.47 lb/
gal at 28°F (the mean annual temperature for Fairbanks in 1970).
Other data on the sample are given in Table 1.

Excellent agreement was obtained between the experimentally
determined value for hydrogen content, 13.51%, and the value
calculated with the Bureau of Standards equation,, 13.54%[1]. The
only variable parameter in the Bureau of Standards equation is
specific gravity:

wt % H = 26 - [15 x (spec gravity of the fuel at
        60°/60°F)]

As Mr. Wiseman pointed out in your telephone conversation we are
currently changing the source of our Alaska Stove Oil. Most

---

1 "Thermal Properties of Petroleum Products", U. S. Dept. of
   Commerce, Bureau of Standards Miscellaneous Publication No. 97
   (11/9/1929). This equation gives the same numbers as the
   nomograph you received from Esso.

properties of stove oils from the old and new source are similar but the new fuel is lighter and will typically have an API gravity at 60°F of 41.6° (0.8174 spec grav at 60°/60°F or 6.806 pounds/gal at 60°F)[2]. From the equation we can calculate the hydrogen content of typical fuel from the new source:

$$wt\% \ H = 26 - 15 \ (0.8174) = 13.74 \ wt \ \% \ H$$

From the known hydrogen content of water we can now determine the number of pounds of water vapor formed per pound of fuel burned:

$$\frac{0.1374 \ lb \ H}{lb \ fuel} \ \times \ \frac{18.02 \ lb \ H_2O}{2.02 \ lb \ H_2} \ = 1.23 \ lb \ H_2O/lb \ fuel$$

or 8.49 lb H_2O/gal fuel at 60°F. For your convenience we have calculated the number of pounds of water formed upon combustion of one pound of fuel for fuels within a range of API gravities. The range corresponds to kerosines, #1 fuels and light #2 fuels. The results are given on the attached graph.

We assumed that all hydrogen present is oxidized to water vapor. In general this is a valid assumption even in cases where combustion is not 100% efficient (i.e., not all carbon is converted to carbon dioxide).
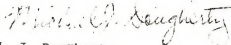
If we can be of any further assistance to you please advise us.

---

[2] By definition:

$$API \ gravity \ at \ 60°F = \frac{141.5°}{spec \ grav \ 60°/60°F} - 131.5°$$

Very truly yours,

M. J. Dougherty
Research Chemist
Products Research

MJD/hw
Attachments
cc: R. J. Askevold
    E. R. Friess, UOC
    H. C. Huffman
    G. E. Irish
    R. M. Matsuo
    E. L. Wiseman

    L. E. Leonard, Jr. (University of Alaska)
    W. C. Stringer (Bureau of Land Management, Colorado)

API Gravity of Fuel vs. Water Vapor Formed by Combustion

Table 1

Properties of Union Stove Oil Sample
Received from the Bureau of Land Management Fire Control
Station at Fairbanks, Alaska

Date Received - February 28, 1972

Lab Book Sample No. - Y1122

Gravity, °API - 38.8

ASTM D-84 Distillation, °F
  Initial Boiling Point - 326
  Temp at % Recovered:

5	358
10	370
20	388
30	402
40	416
50	431
60	445
70	462
80	480
90	512
95	550
End Point	584
% Recovered	98.5

Carbon Content, wt % - 86.5 (average of duplicates)

Hydrogen Content, wt % - 13.51 (average of four replicates)

Sulfur, ppm - 206 (average of duplicates)

PROD-72-331
3-14-72

APPENDIX III

O & M MANUFACTURING COMPANY

PRESSURE DROP CALCULATIONS AND SPECIFICATIONS

FOR THE HEAT EXCHANGER USED ON PROJECT

(Calculations retyped from O & M original)
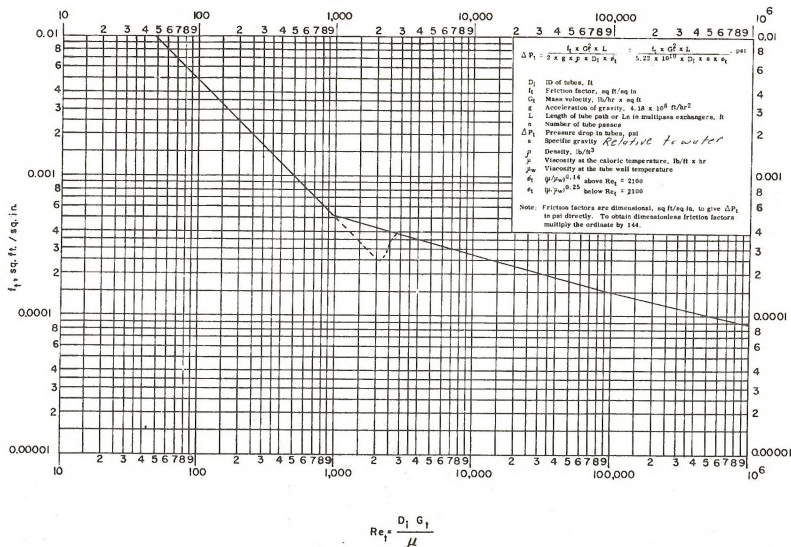
Fig. 19 Tube Side Friction Factors

[Adapted from Sieder and Tate and Kern, "Process Heat Transfer" by permission McGraw-Hill Book Co.]

$$Re_t = \frac{D_i \, G_t}{\mu}$$

The chart contains the following equation and legend:

$$\Delta P_t = \frac{f_t \times G_t^2 \times L}{2 \times g \times \rho \times D_i \times \phi_t} = \frac{f_t \times G_t^2 \times L}{5.22 \times 10^{10} \times D_i \times s \times \phi_t} \cdots \text{psi}$$

D_i = ID of tubes, ft
f_t = Friction factor, sq ft/sq in
G_t = Mass velocity, lb/hr x sq ft
g = Acceleration of gravity, 4.18 x 10^8 ft/hr^2
L = Length of tube path or Ln in multipass exchangers, ft
n = Number of tube passes
ΔP_t = Pressure drop in tubes, psi
s = Specific gravity Relative to water
ρ = Density, lb/ft^3
μ = Viscosity at the caloric temperature, lb/ft x hr
μ_w = Viscosity at the tube wall temperature
ϕ_t = $(\mu/\mu_w)^{0.14}$ above Re$_t$ ≥ 2100
ϕ_t = $(\mu/\mu_w)^{0.25}$ below Re$_t$ ≤ 2100

Note: Friction factors are dimensional, sq ft/sq in. to give ΔP_t in psi directly. To obtain dimensionless friction factors multiply the ordinate by 144.

PO# __52500-CT2-46__ Job No. __80079__

HGC-60-815 (Model #)

(Calculations retyped from O & M original)

U.S. Government
Bureau of Land Management

Exhaust Gas Cooler

Given 541cfm air suction at 433' elevation (use 32°F)

air density = .0794 #/ft^3
air flow = 2577 #/hr.

Given 21.2 gal./hr. of #2 diesel fuel

sp. gr. = .95
fuel flow = 167.7 #/hr.

Water suction if air saturated	=	9.8 #/hr.
Water formed during combustion	=	209.6 "
Water vapor to cooler	=	219.4 "
Water content of cooler discharge	=	11.4 "
Water vapor condensed	=	208.0 #/hr.

Component	To Cooler		From Cooler	
	#/hr.	#Moles	#/hr.	#Moles
CO_2	521.5	11.85	521.5	11.85
O_2	73.0	2.28	73.0	2.28
CO	18.1	.65	18.1	.65
H_2	0.8	.4	.8	.4
N_2	2131.3	76.07	2131.3	76.07
	2744.7		2744.7	
H_2O	219.4	11.8	11.4	.63
	2964.1	103.44	2756.1	91.88
	Mole wt. = 28.65		Mole wt. = 30.	

Suction volume = $\dfrac{2964.1}{(4.5)(\frac{28.65}{29})}$ $(\frac{960}{530})$ = 1207.67 cfm

Discharge Volume = $\dfrac{2756.1}{(4.5)(\frac{30}{29})}$ $(\frac{495}{530})$ = 552.96 cfm

liquid 208$(\dfrac{.01602ft^3/\#}{60 \text{ min.}})$ = $\dfrac{.05}{553.01}$ ft^3/min.

Gas to Cooler

Density at 500°F & 14.7 P_{sia} $T_{avg} = 268°$

$$\rho = .075 \, (\frac{28.65}{29})(\frac{530}{728}) = .0539$$

Sp. Gr. (relative to water) = .000864

$$G = \frac{2964.1 \ \#/hr.}{352 \ tubes} = 8.42 \ \#/hr./tube$$

$$G_t = \frac{8.42}{.000134} = 6284 \ \#/hr./ft^2$$

$$R_e = \frac{.04125(6284)}{2.42(.0215)} = 4982$$

f = .000327

$$\Delta P_{Friction} = \frac{(.000327)(6284)^2(12)(1)}{(5.22 \times 10^{10})(.04125)(.000864)(1.03)} = .081 \ psi$$

Back Pressure Due to Change in Flow Area & Contraction (Assumed initial pressure of 14.7 P_{sia})

Inlet   16" X 10" Concentric Reducer   ID = 15.25"

Outlet   12" sch 20 Pipe

Inlet Volume 1207.67 cfm

Outlet Volume 553.01 cfm

$$P_2 = 14.7 \, (\frac{1.268ft^2}{.8184ft^2})(\frac{553.01}{1207.67}) = 10.4$$

Open to atmosphere at 14.45 P_{sia}

Use opening to separator 10" sch 20 pipe
ID = 10.25

$$P_2 = 14.7 \, (\frac{1.268ft^2}{.573ft^2})(\frac{553.01}{1207.67}) = 14.9$$

$$\frac{-14.7}{.2} \ psi$$

.281 psi (2.31)(12) = 7.79" Maximum
Due to drawing effect of atmospheric pressure probable back
pressure 2 → 3"

O & M MANUFACTURING COMPANY
Houston, Texas

Hot Gas Cooler Specification Data Sheet

Purchaser	U. S. Department of the Interior	Proposal	
Ref. No.	P. O. #52500-CT2-29 (DSC-72-46)	Date	July 26, 1971
Model	HGC - 60 - 815	Page	1 of 1

PERFORMANCE

Fluid	Flue Gas
Flow	2744 #/hr.
Temperature In, °F	500
Temperature Out, °F	35
Water Suction, #/hr.	9.8 (541 CFM of saturated air @ 32°F)
Water Formed, #/hr.	209.6
Water Condensed, #/hr.	208
Water Not Condensed, #/hr.	11.4 @ 35°F
Pressure, PSI	
Pressure Drop, PSI	Nil
Total Heat Load, BTU/hr.	587,287
LMTD	161.9
Overall Rate	5.3
Fouling Factor	.01
Tube Surface, Sq. Ft.	684
Sections, No. of	1
Passes, No. of	1
Design Temp., °F	750
Design Press., PSIG	15
Test Press., PSIG	25
Inlet Nozzle	16" Dresser
Outlet Nozzle	10" Dresser
Tubes, OD x BWG	5/8 x 16
Material	SA-334 Gr. 1
No./Section, Length	352 - 12'
No. Rows	8
Layout	1-1/4" Triangular Pitch
Fins	None
Headers, Type	3/4" Plate
Material	SA-516 Gr. 70
Tanks	Welded Inlet, Bolt on Outlet
Material	SA-516 Gr. 70
Plugs	None
Accelerators	None
ASME Code Stamp	No

Total SCFM	27,600	Temp. In, °F	0	Temp. Out, °F	20	Elev.	435

MECHANICAL EQUIPMENT

No. Fans 2 HP/Fan 2.7 RPM 1200 No. Blades 6 Material Al. Model 42 - 4
Fan Make Western   Draft Type F   No. Bays 2
Drive   Direct with 2, 3 HP, 3/60, 220v, 1200 RPM, TEFC electric motors with
ball bearings permanently lubricated with low temperature silicone grease
Paint   O & M Standard   Assembly   Packaged
Remarks: See drawing for shutter and actuator details
O & M Job #30079
O & M Dwg. #8595